Adjustable Speed—A Tool for Saving Energy Losses in Pumps, Fans, Blowers, and Compressors

HERBERT N. HICKOK, SENIOR MEMBER, IEEE

Abstract—**Petroleum and chemical plants today are effectively cutting energy use by thermally, electrically, and mechanically reducing losses in their process equipment. In rotating equipment such as pumps, fans, compressors, and blowers, much mechanical or fluid energy is being dissipated or thrown away by throttling valves, dampers, and adjustable guide vanes. Adjustable speed is an increasingly used feature in the search to reduce losses. Fluid mechanics is reviewed in lay terms in an effort to illustrate where losses may be found and how to determine how much may be saved by converting to or using adjustable speed. Other potential ways that adjustable speed may be used for process savings are also discussed.**

INTRODUCTION

IN TODAY'S efforts to save energy in refineries, petrochems, and chemical plants, much is being done both thermally and electrically to save energy. Following the original oil crunch, energy was saved primarily to save fuel. Today with the job well along and the petroleum demand down, the utilization factor of plant capacity has dropped from 90–95 percent down to around 65 percent, though it is now swinging up. The turndown of a refinery with many normal fixed losses means that it operates much less efficiently. In the present competitive situation with reduced demand, there is much more incentive to find and eliminate energy losses to return to a more profitable operating condition. Since 1972, at the beginning of the oil crunch, refineries have reduced the amount of energy they consume to process a barrel of crude into products by an average of around 30 percent. By contrast, the steel industry has saved approximately ten percent (Japanese steel industry, 18 percent), the paper industry ten percent, the cement industry 15 percent, and the aluminum industry 20 percent. All industries, of course, had increased energy use because of pollution equipment. The petroleum and petrochemical industry, with 40–50 percent of its product manufacturing cost in energy, is expected to continue to improve to around 40 percent in the next several years as shown in Fig. 1. Much of these savings have been in the thermal area which originally gave the fastest payouts per dollar invested. As electrical power costs continue to climb, however, more and more attention is being paid to saving electrical losses, principally by turning to cogeneration, more efficient distribution equipment, high-efficiency motors, and adjustable speed drives.

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The author is with the General Electric Company, P.O. Box 4367, Houston, TX 77210.

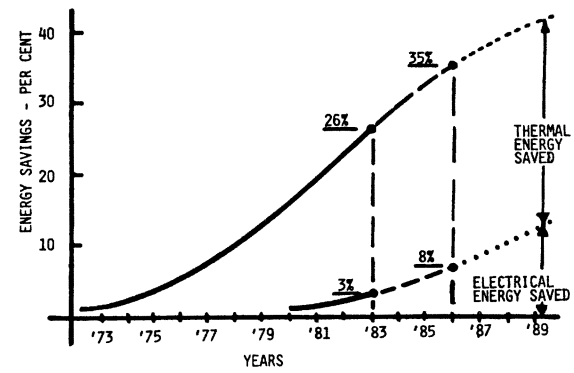

Fig. 1. Approximate energy savings in petroleum and chemical plants since 1973.

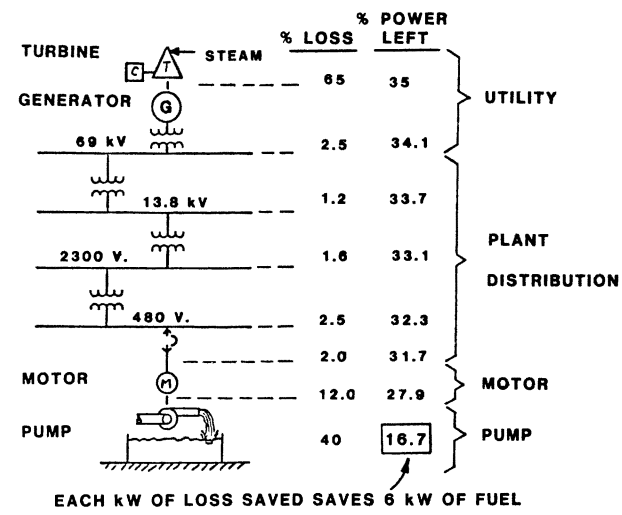

Fig. 2. Flow of energy and its typical losses through power system.

Fig. 2 gives us a picture of the flow of energy through a power system starting from fuel energy input into, say, a boiler down to actual work done by utilization equipment. Obviously, the big losses to be saved are thermal losses at the front end of the energy system, and this is the reason so much attention is being paid to combined cycle and cogeneration power plants. If we put 100 kW of fuel energy into the system, note that we normally get only 15–20 kW of useful work out at the utilization end. We might wonder why we should spend much effort on motors and their driven equipment when the large losses are at the front end. The answer comes quickly, however, when it is realized that, for every kilowatt of loss saved in the electrical utilization equipment or process drives, 6 kW of fuel energy has been saved on the front end of the

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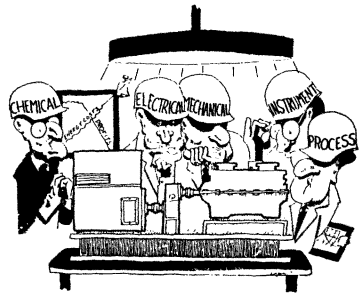

Fig. 3. Many plants today use energy teams to find and evaluate process losses.

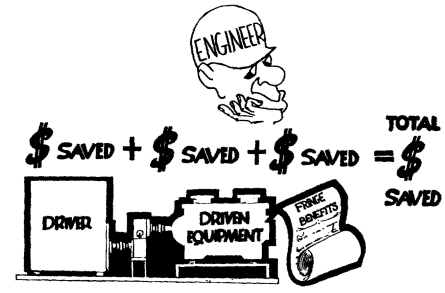

Fig. 4. In process drive, energy savings may be made in driver and/or driven equipment and/or fringe benefits. It is sum of any or all of these.

system. This point brings into sharper focus the need to find and save losses in process drives.

Process drive losses are not always easy to find. Like prospecting for oil, you may have to do a lot of drilling before you hit a real gusher. They are pretty much silent and concealed in mechanical, electrical, fluid, and chemical characteristics of a process and its equipment. They must be dug for in characteristic curves, fluid mechanics, process theory, work equations, and electrical circuits. Because a number of disciplines are involved, the trend today is to put together an energy saving team. Mechanical, chemical, electrical, process, and instrumentation engineers are assembled to proceed through the plant on a systematic energy audit. Their objective is to find, decide how to save, evaluate, and report to management their recommendations on energy savings projects. It is also their job at times to carry them out. Energy saving today is becoming a team operation. The cartoon in Fig. 3 shows a theoretical team in operation. The time they spend per project is expensive, and so it is important that they learn the best places to look for losses and the tools with which they have to work.

Adjustable speed is turning out to be one of the best tools for saving energy. How to use it to reduce losses is the subject of this paper. It is applicable in both new and retrofit projects.

ADJUSTABLE SPEED ENERGY SAVINGS

Adjustable speed saves energy in several different ways. First is the saving in one type of adjustable speed driver over another when the decision has been made to use adjustable speed. Typical would be the saving in efficiency in a comparison of an adjustable frequency drive over that with a hydraulic coupling or a steam turbine. Second, potential savings are found on the other side of the coupling in the driven equipment, those that are typical by using adjustable speed in place of throttling or damper control. Then possible fringe benefits exist in one or more features that adjustable speed brings with it as part of its package. On a given process drive (Fig. 4), the actual savings are a total of all three of these. The format for an evaluation or payout follows this general form:

a) savings in driver $ _____
b) savings in driven equipment (fan, pump, compressor or blower) $ _____
c) other fringe benefits $ _____

TOTAL SAVINGS FOR PROJECT $ _____

In some cases total savings may be found in only one of the three categories. In some it can be in all three. Fringe benefits, as mentioned later, may be hard to evaluate in dollars, yet in some cases it may be because of them alone that adjustable speed is chosen for a drive. To the gross savings the time value of money must be considered, though it is beyond the scope of this paper.

Many recent good technical papers are available which discuss and compare different types of adjustable speed drivers. Because of this and limited space, this paper discusses categories b) and c) only, namely savings in the driven equipment and in fringe benefits. Though the paper will not discuss different drivers for process rotating equipment, it will be worthwhile to review (Table I) what choices are available in adjustable speed drivers. The sizes and practical speeds listed are those generally found in refining and petrochemical process drivers, not necessarily the maximum available.

Finding and Evaluating Losses in Driven Equipment

Perhaps the greatest savings with adjustable speed are those made by using adjustable speed in conjunction with the design characteristics of pumps, fans, compressors, and blowers to improve performance (category b)). Although the largest in amount of savings, they vary considerably from near nothing to real gold mines. Doing homework here pays off, for some have rushed to convert to adjustable speed without looking into what happens inside the driven equipment only to find next to no savings in energy when the conversion is finished.

To understand how losses are eliminated, a short review of fluid mechanics is needed. Though pumps are discussed later, the head–flow (H–Q) and system curves of pumps, fans, compressors, and blowers all work in much the same way. Energy savings calculations and formats follow in the same manner.

PUMPS

A. Head-Flow Curves

Typical characteristic data of a pump as provided by a manufacturer are shown in Fig. 5. It consists of a curve showing how flow of a fluid varies as a function of pressure and friction. It must also include an efficiency curve showing how efficiency varies as a function of head and flow. The H–Q curve provided is for a given specific speed, fluid pumped, density, viscosity, and impeller diameter and width. If any of these vary, the pump curve must be adjusted. Fig. 6 shows

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TABLE I
ADJUSTABLE SPEED DRIVERS AVAILABLE IN PETROCHEMICAL SERVICE

Type of Driver	Speed Range Percent	r/min (without gear)	Range (hp)
Steam turbine	0-100	6000	100-60 000+
Gas turbine (2 shaft)	70-100	8000	1000-60 000+
Hydraulic coupling	0-100	3600	100-20 000+
Direct current			
Low hp	0-100	1800	1-200
High hp	0-100	800	300-3500
Adjustable frequency			
Low voltage (600 V max.)			
Transistorized	10-100	3600	1-25
Square wave	10-100	3600	1-1500
High voltage (6600 V max.)			
Cycloconverter	0-100	20	10-15 000
Kramerstat	80-100	900	500-5 000
Load commutated (low speed)	10-100	1200	500-60 000+
Load commutated (high speed gearless)	10-100	7500	2000-60 000+

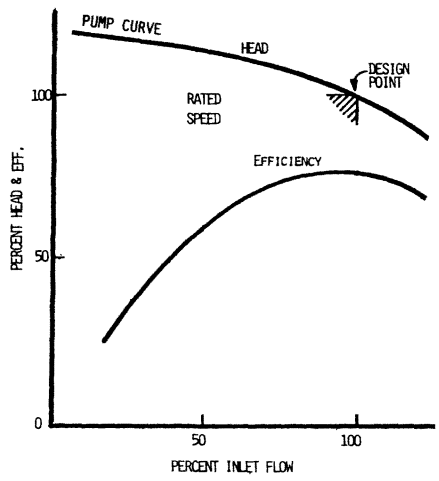

Fig. 5. Typical head-flow (H-Q) and efficiency curve of specific pump at rated speed.

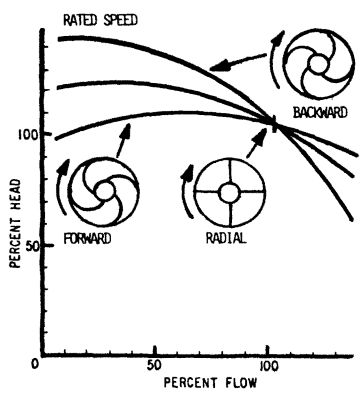

Fig. 6. Pump impeller, by having forward, radial, or backward leaning impeller vanes, can change shape of H-Q curve.

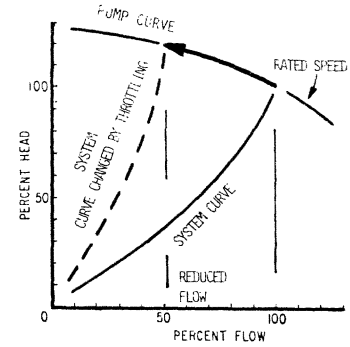

Fig. 7. Typical pump H-Q curve with system curve added. Artificially adding friction by throttling changes system curve and thus reduces pump flow.

how forward, radial, or backward impeller vanes in the pump typically change the curve also.

A specific pump installation in a process has a system H-Q curve. The flow of fluid varies with the amount of fluid friction, discharge pressure, and change in elevation. Fluid friction is a function of pipe diameter, length, number of valves and fittings, smoothness of inside finish, and density and viscosity of fluid pumped. The system resistance curve must be determined for each pump (or blower, fan, or compressor) and product to be investigated. With an existing installation for retrofit consideration, it can possibly be measured by varying flow and recording output pressure. A number of textbooks, technical papers, and vendor aids are available to help determine the curve of fluid friction for a planned process installation. Fig 7 shows a typical pump and system H-Q curves. Fig. 8 shows the effect of a change in product density.

With a fixed speed pump, flow can only vary as a function of pressure along its original H-Q curve. The actual amount of flow is the point at which the system resistance and pump curves cross. To decrease flow, additional resistance is added to system resistance by means of a throttling valve which forces the pump to operate at a point back along its curve at the desired flow rate. Fig. 7 indicates how this is accomplished. The differential in pressure between the system curve and the pump curve is a measure of the throttling loss which represents energy dissipated by the throttling valve in order to achieve a lower desired flow. In Fig. 7 low or zero flow was achieved at zero pressure, indicating the pump operated against no back pressure. If the pump had to raise its fluid output into an elevated tank or pump against a fixed head, then the system head curve would look more like that in Fig. 9. If the pump had to operate into a constant pressure manifold, then the system curve would look like that in Fig. 10. Note that as the system back pressure goes up, the lower the throttling loss becomes, and this gives a clue as to where to find the greatest potential savings in energy.

The horsepower input required by a pump is

$$\text{hp} = \frac{(\text{gal/min}) \times (\text{ft head})\ (\text{SG})}{3960 \times \text{Eff}}.$$

where SG is specific gravity. Other forms of the equation are

$$\text{hp} = \frac{(\text{BBL/hr.}) \times (\text{lb/in}^2)}{2450 \times \text{Eff}}$$

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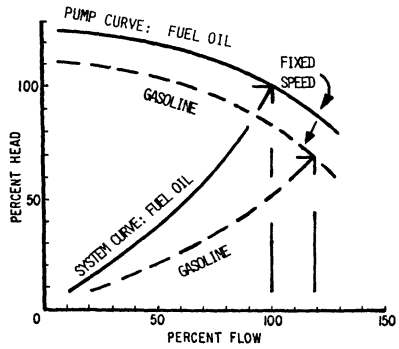

Fig. 8. Products through same pump of different density (specific gravity) change both pump and system curves.

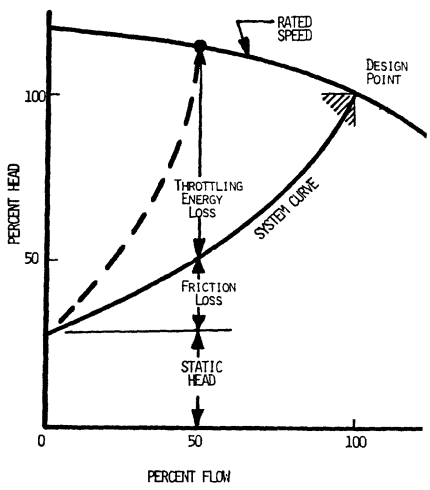

Fig. 9. As system back pressure increases, higher system curve starts at no flow.

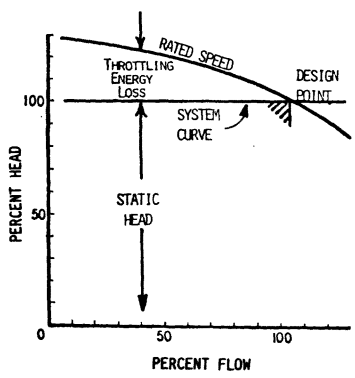

Fig. 10. Pumping at constant pressure provides horizontal system curve, and energy savings are minimized.

or

$$\text{hp} = \frac{(\text{gal/min}) \times (\text{lb/in}^2)}{1714 \times \text{Eff}} .$$

Horsepower lost by throttling for a given flow is a measure of the difference in pressure as indicated by the height of the throttling loss line. Since pump efficiency changes as a function of flow, it must be changed also. Many vendors fortunately will provide a horsepower–flow curve with their head–flow and efficiency curves, and this is a big help in getting to the answer in an evaluation of losses.

Another way to change the flow of a pump is to change

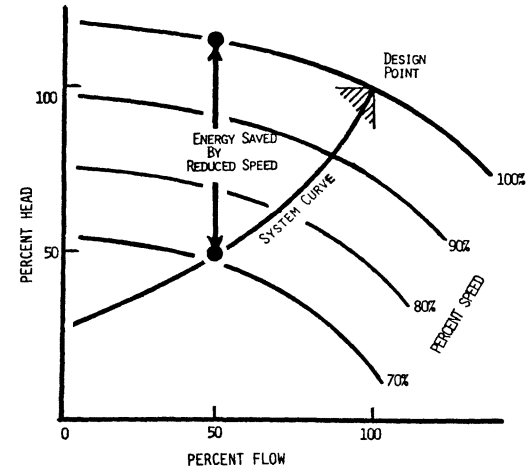

Fig. 11. Family of pump H–Q curves at various speed can be constructed from original fixed-speed curve.

speed. The general pump–speed relationships are as follows:

- Q (flow) varies as S (speed),
- H (head) varies as S^2,
- HP varies as S^3.

The relation of these parameters holds for a given point on the pump H–Q curve. Given the original constant speed H–Q curve of a pump, a family of curves at various speeds can be constructed as shown in Fig. 11. For instance, a curve with a ten-percent decrease in speed (90-percent speed) can be cnstructed whose H–Q coordinates would have their head values at 81 percent of their original, while their flow values would be 90 percent. The hp values would be down by 72.9 percent (0.90^3) adjusted by any change in efficiency. As it turns out, efficiency generally holds close to being constant as you change speed and come down the system head line over about the upper half of the flow range, 50–100 percent. Fig. 11 shows the same curve as Fig. 9 except the throttling loss has been eliminated by reducing the flow by decreasing speed. The pump operating point follows down the system head curve to a new lower flow, and throttling losses are eliminated. *This is the main key to saving energy by adjustable speed.* It applies to almost all kinds of rotating pumps except the positive displacement type which are beyond the scope of this paper.

There are various types of rotating pump designs. In types of pump flow they are classified as follows.

- *Radial:* Flow is turned 90° as it goes through the pump. These are centrifugal pumps and are the most common.
- *Mixed:* Flow is turned less than 90°, generally around 45°, and they are turbine pumps. Frequently, they are used with multiple stages.
- *Axial:* Flow is straight through and along the shaft. They are propeller pumps.

Fig. 12 shows the variation in the H–Q curves of different types of pumps. Since throttling energy saved is the difference between the pump and system curves, the steeper the slope of the pump curve, the greater the savings. This is another clue to look for in sorting out which potential applications offer the greatest saving. Bear in mind that a wide range of curve shapes exists for each of these types of pumps.

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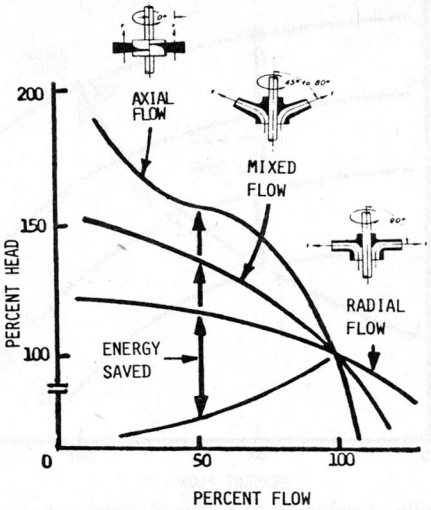

Fig. 12. Characteristic H–Q curves are different for radial, mixed, and axial flow pumps.

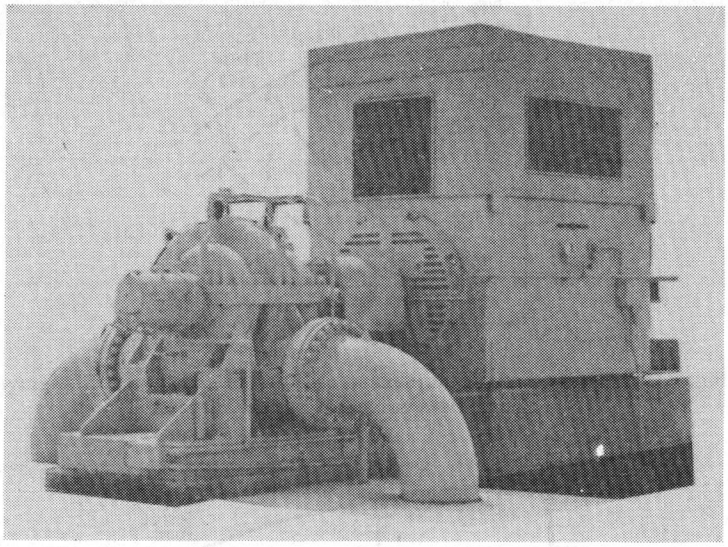

Fig. 13. Typical outdoor pump drive similar to that in example.

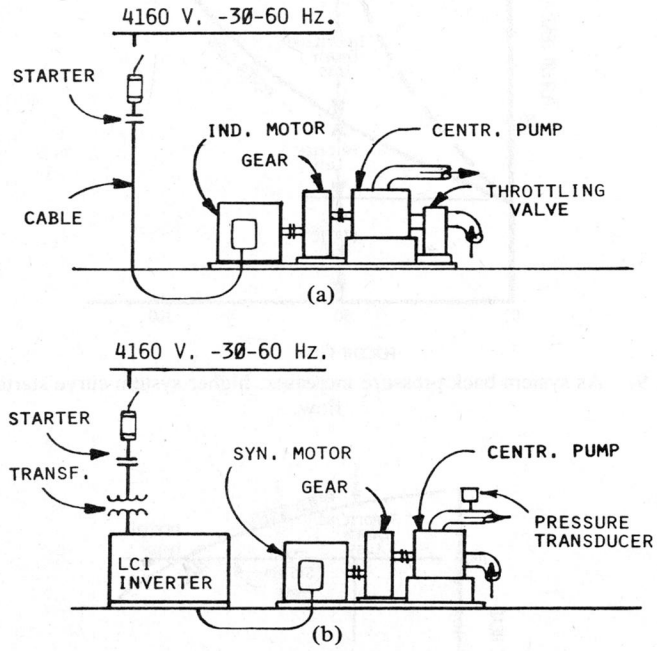

Fig. 14. Schematic arrangement. (a) Fixed speed. (b) Adjustable speed.

B. Duty Cycle

A second factor of real importance is the amount of time that a pump operates at less than its rated point. Pumps are chosen with some margin to begin with. As pipe, valves, fittings, and pumps age, their interior surfaces roughen up and corrode and their friction to flow increases. The engineer who picks the pump still wants rated output towards the end of its life though the system friction curve will become steeper with time. When the installation is new, the pump must be throttled back at its system rated flow point to allow a margin in pump size selection and for future system friction increase. At the start, then, a pump at rated process throughput operates with some throttling. The same is usually true for fans, compressors, and blowers.

Many things can cause a variation in flow or turn down. Market demand may require a process cutback. A change in ambient temperature, specific gravity, the amount of distillation reboil or reflux, or a change in product are but a few. The greater the time a pump operates at reduced flow and the greater the reduction in flow, the more the losses in energy add up by throttling. At times, the reduction in flow is a planned fixed amount for a prolonged period of time such as, say, several years on a planned product change or output. If that is the case, it may be practical to stay with fixed speed but change the diameter of the pump impeller. For a given pump, the manufacturer may make available a series of impellers of different diameters for this purpose. In many cases, however, the amount of turndown of a pump varies hourly or daily for a range of desired flows. In many cases it is controlled by a pressure or flow controller, and the changes come automatically.

In order to evaluate a potential saving with adjustable speed over fixed speed, an estimated duty cycle must be made. If the retrofit of a given pump is the project, then it likely can be determined more accurately by actual operation. If the installation is new, then the duty cycle must be estimated from projected and planned operation. Since the savings per year are the answer being sought, breaking up the planned hours of operation per year at various reductions in flow is the easy way to do this. Determining energy saved at each level of reduction times the time in hours/year all added together gives the total saving.

C. Example

The following example, comparing fixed speed with throttling versus adjustable speed for a 2500-hp centrifugal pump drive operation, similar to that in Fig. 13, illustrates the format for determining savings and ultimate payout. Fig. 14 shows the schematic arrangements of the two systems, and Table II gives some basic data. The head–flow and hp curves are given in Fig. 15. The differential cost comparison is found in Table III.

At 85-percent kilowatt load,

$$\text{power saved} = 2200 - 2060 = 140 \text{ hp}$$

$$= \frac{2200 \times 0.746}{0.955 \times 0.98} - \frac{2060 \times 0.746}{0.945 \times 0.98}$$

$$= 1754 - 1659 = 95 \text{ kW}.$$

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TABLE II
DATA

	Fixed Speed	Adjustable Speed
Motor		
Horsepower	2500	2500
Type	Induction	Br. Synchronous
Power factor	0.9	1.0
Voltage, 13.8 kV	2400 V	2400 V
Efficiency	95.5 percent	(with inverter)
Cost (installed)	$39 776	$54 808
Enclosure	WP IV	WP II
Inverter (load commutated)		
Efficiency		
(inc. motor)		94.5
Cost (installed)		$200 000
Gear		
1800/10.950 r/min	continuous duty	continuous duty
Efficiency	98 percent	98 percent
Duty cycle (estimated)		
6000 h/yr. @ 85 percent flow		
2000 h/yr. @ 75 percent flow		

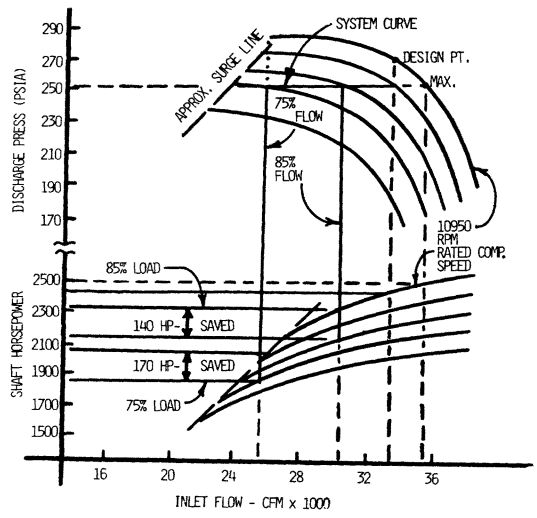

Fig. 15. Head–flow and hp curves.

TABLE III
DIFFERENTIAL COST COMPARISON

	Fixed Speed	Adjustable Speed
1) Starter	$ 9500	$ 8500
2) Isolation/stepdown 2500 kVA Transformer	$26 000	$ 25 000
3) Motor (installed) (Ind.-Synchronous)	$39 776	$ 54 808
4) Inverter (Installed—inc. floor space)	——	$200 192
5) Gear stepup unit—same both cases (1800–10 950 r/min)	——	——
6) Suction throttling valve and air system	$12 000	——
7) Constant press control components	——	$ 2000
Total cost	$87 276	$290 500
Incremental difference favor fixed speed drive	$213 225	

At 75-percent kilowatt load,

$$\text{power saved} = 1950 - 1780 = 170 \text{ hp}$$

$$= \frac{1950 \times 0.746}{0.955 \times 0.98} - \frac{1780 \times 0.746}{0.945 \times 0.98}$$

$$= 1554 - 1433 = 121 \text{ kW}.$$

The power cost at present is $0.035/kWh (doubling every six years). In 20 years, the cost is projected to be $0.126/kWh, giving an average power cost of $0.081 for 20 years.

The kilowatt hours saved are tabulated as follows:

ft³/min (percent)	kW Saved	Hours	kWh
85	95	6000	570 000
75	121	2000	242 000
	kWh saved/year =		812 000

This gives power savings/year as

$$812\,000 \times 0.081 = \$65\,772$$

(does not include savings of 95 kW in demand charge). This in turn gives

$$\text{payout of savings} = \frac{\text{additional investment}}{\text{savings/year}}$$

$$= \frac{\$213\,224}{\$65\,772} = 3.24 \text{ years}$$

and

$$\text{gross life savings (over 20 years)} = \$65\,772 \times 20$$

$$= \$1\,315\,440$$

or nearly $1 500 000. The overall cash flow is depicted in Fig. 16.

The foregoing illustration is not a particularly good one because, as mentioned earlier, it has a high constant back pressure to pump against, and the centrifugal pump curve is rather flat over the operating range. Even so, the 3-yr payout in almost a worst-case condition is not out of consideration. A 20-lb decrease in back pressure would cut its payout time considerably.

D. Factors Affecting Data

Data for pumps under consideration for adjustable speed must be specific to be valid. Among data and factors affecting the evaluation are

1) exact H–Q curve of pump,
2) H–Q curve good for only one:
   a) viscosity
   b) density of fluid
   c) temperature as it effects viscosity
   d) shape of impeller
   e) diameter of impeller
   f) width of impeller,

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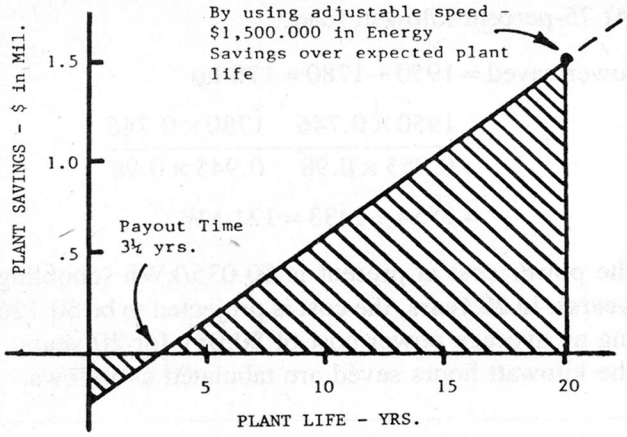

Fig. 16. Cash flow.

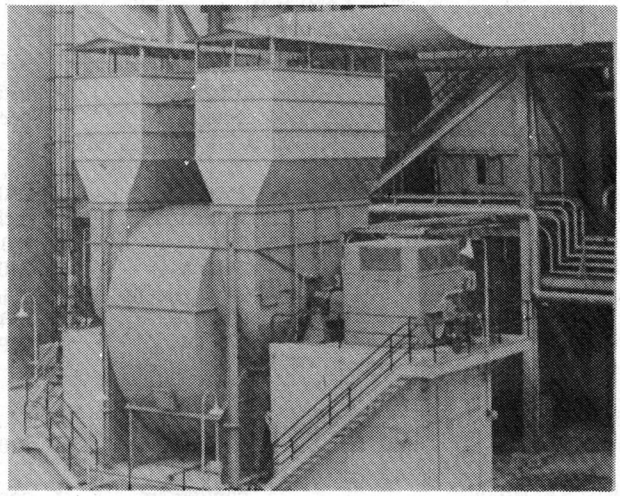

Fig. 17. Blowers and fans are typically good candidates for saving energy by adjustable speed.

3) system H–Q curve,
4) number of pump stages,
5) parallel operation of pumps,
6) future changes in system curve,
7) cost of energy and demand charges and their future changes over payout period,
8) predicted life of installation, and
9) duty cycle of the pump.

FANS AND BLOWERS

A. Head–Flow–HP Relationship

The fluid mechanics of fans and blowers is almost the same as pumps except that Q (flow) is in terms of ft^3/min and H (head) is generally in terms of inches of H_2O. Power to the input shaft is

$$\text{hp} = \frac{P \times Q}{6356 \times \eta}$$

where

P pressure, in of H_2O,
Q flow, ft^3/min,
η fan or blower efficiency, percent.

The equation holds true as long as little or no work of compression is done on the fluid, say air. If output pressure is high enough to cause compression of more than a few percent (say, ten percent maximum), then the compressor equation given later is needed. Most fans and blowers have relatively little static back pressure to pump against. Because of this and the fact that their H–Q curves slope downward sharply, they are good candidates for energy saving projects. About 65 percent of the large installations to date using adjustable speed have been on blowers. Fig. 17 shows a typical installation.

As fixed speed drives, blowers for the most part have had their flow controlled by outlet damper. It is the least complicated in construction. Fig. 18(a) shows the pressure–flow and horsepower flow relationship which is much the same as for throttling on a centrifugal pump. System head resistance is added by the damper to force operation back up the head–flow characteristic to a desired reduced flow. Though the flow is reduced considerably, there is a relatively small reduction in horsepower.

Another more sophisticated type of control is with inlet guide vanes. Here adjustable vanes slide across the inlet and guide and restrict air to the blower to control flow. This holds back or reduces the weight of air that can enter, thus reducing the work required. Fig. 18(b) shows that a new and steeper head–flow and horsepower–flow curve is formed since in effect the density has changed. Keep in mind that in fluid mechanics a given head–flow characteristic changes as density of the fluid (air in this case) changes. The horsepower curves in Fig. 18(b) show that for the same reduction in flow as in Fig. 18(a) a greater savings in horsepower is attained than for the comparable outlet damper case.

Fig. 18(c) shows how the head–flow and horsepower–flow curves change as adjustable speed is used on the same blower. The head–flow curve is reduced in the same way as was mentioned for a pump, namely, that flow reduces linearly and pressure reduces as the square with speed reduction. Assuming the same reduction in flow or capacity as in Fig. 18(a) and (b), the horsepower required is now substantially less. This is the reason that fans or blowers are such good candidates for adjustable speed operation.

Fig. 19 shows a typical comparison of various methods of control and the relative difference in power required as a function of flow. The axial fan, which is included in addition to the methods described earlier, is one in which the blades change pitch angle while in operation to control flow in like manner to that of an aircraft propeller. Its construction is rather complicated to adjust flow in operation and so far has not been used extensively in hydrocarbon processes.

One of the fringe benefits of speed control on blowers or fans is a rather dramatic reduction in noise level as flow (speed) is reduced. Other types of control do not provide much change.

B. Example

An example of the savings in adjustable speed is that of a 400-hp blower in a glass plant. As indicated in Fig. 20, an existing blower drive, after being studied, was retrofitted to adjustable speed by adding an adjustable frequency power module between the motor and its starter. The calulations are not detailed for space reasons, but the basic data for the

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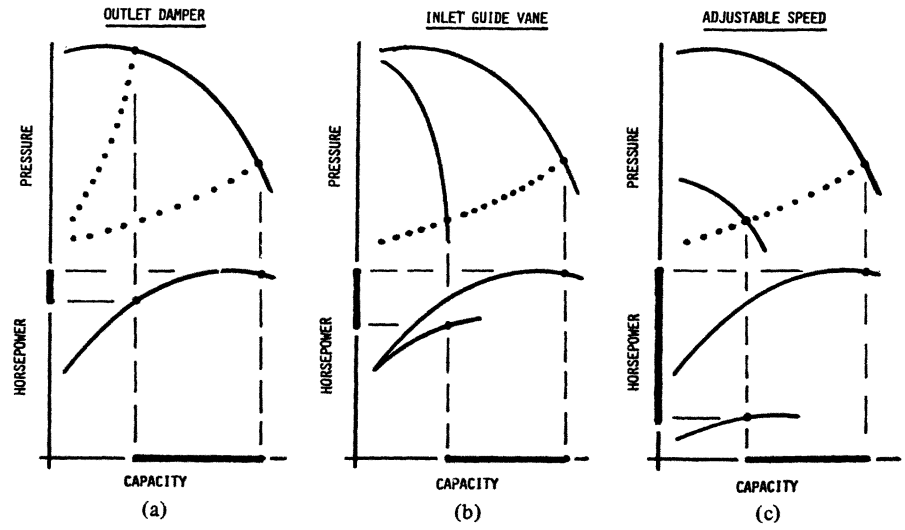

Fig. 18. Relative power saved with different types of flow control on typical blower. (a) Outlet damper. (b) Inlet guide vanes. (c) Adjustable speed.

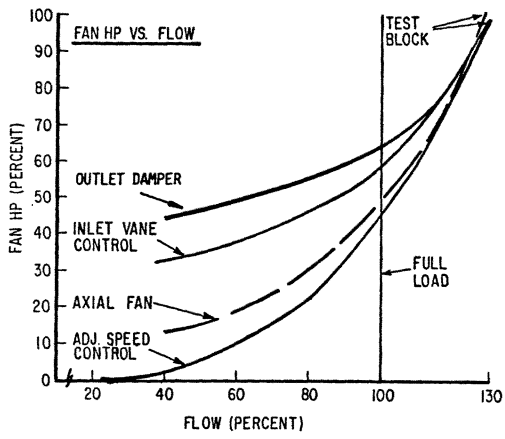

Fig. 19. Comparison for flow verus percent horsepower for various types of blower and fan control.

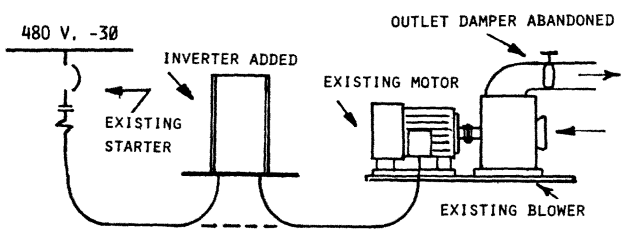

Fig. 20. Energy savings of retrofit blower: mechanical damper versus adjustable speed control.

application are shown:

GLASSPLANT AIR BLOWER	
Fan hp	400
Use	continuous (8760 h/yr)
Duty cycle	85 percent time @ 80 percent flow, 15 percent time @ 95 percent flow
Power cost	3 1/2 ¢/kWh
Inverter cost, installed	$35 000
Energy savings/yr	$47 000
Payout	0.74 years

Its savings of $47 000 a year provided a payout of only nine months. In looking for places to spot energy savings in your plant, fans and blowers should be prime targets. Fin-fan heat exchangers and cooling towers are good places to start.

COMPRESSORS

A. Head-Flow-HP Relationship

Gas compressors are a third area in a refinery or petrochemical plant that offers good potential savings in energy with the use of adjustable speed. They range in size from a few horsepower to nearly 60 000. In reciprocating compressors the savings in adjustable speed are mainly between types of drivers and their efficiencies. In centrifugal compressors, the same type of saving exists, but in general there are greater savings in what happens inside compressors.

The same fluid mechanics laws apply as in pumps and blowers, but they become more complicated when work is done on the gas in compressing it, and because frequently a gas is a mixture of hydrocarbon components, each with its own molecular weight, specific heat, and critical pressure. The equation for horsepower is

$$\mathrm{hp}_p = \frac{\mathrm{WH}}{33\,000\ \eta}$$

$$\mathrm{hp}_p = \frac{\mathrm{W}\left(\dfrac{Z_s+Z_d}{2}\right)\left(\dfrac{1545}{M\omega}\right) T_s \left[\dfrac{r_c^{(n-1)/n}-1}{(n-1)/n}\right]}{33\,000\ \eta}$$

where

hp	polytropic gas horsepower,
W	weight flow in lb/min,
Z_s, Z_d	compressibility factors for suction and discharge press,
M	molecular weight of gas mixture
T	suction temperature in °R,
r	ratio of compression,
η	polytropic efficiency,
r_c	polytropic compression ratio; i.e., P_d/P_s.

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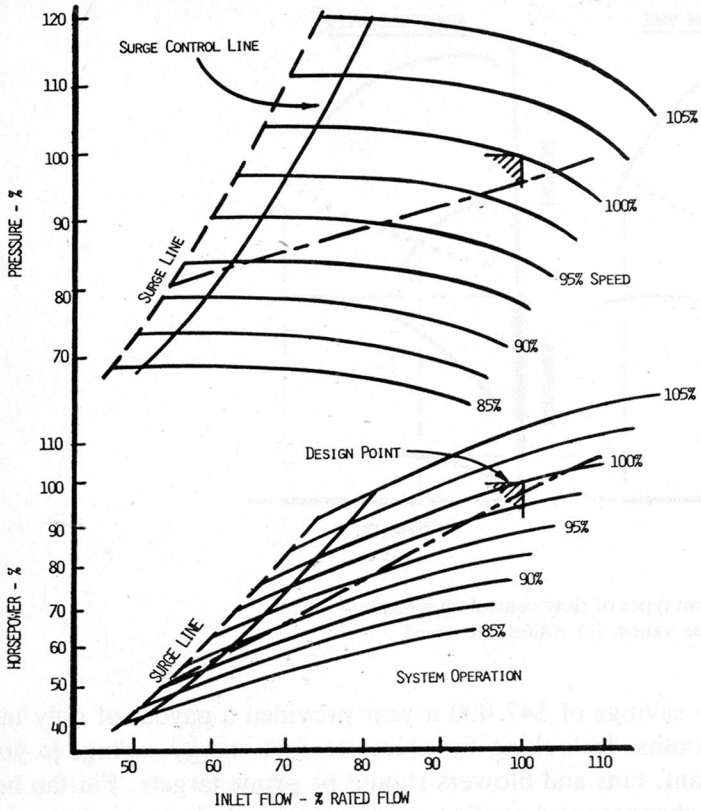

Fig. 21. Typical H–Q and horsepower curves for range of speeds for compressor.

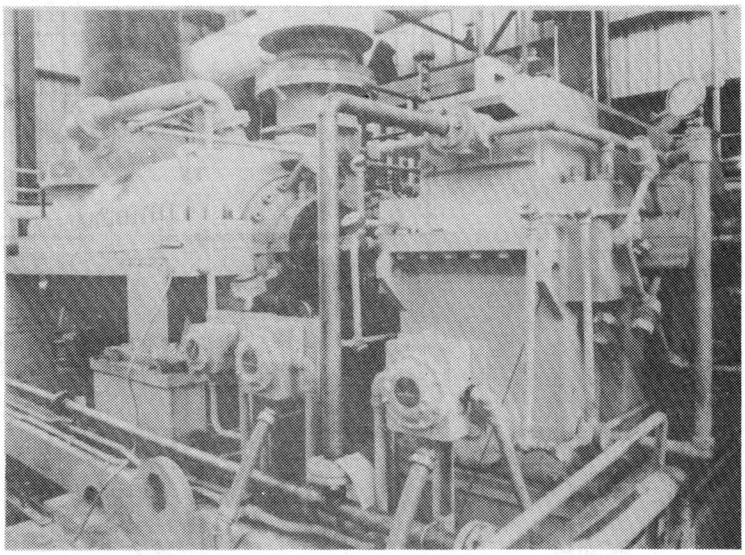

Fig. 22. Typical 10 000-hp compressor.

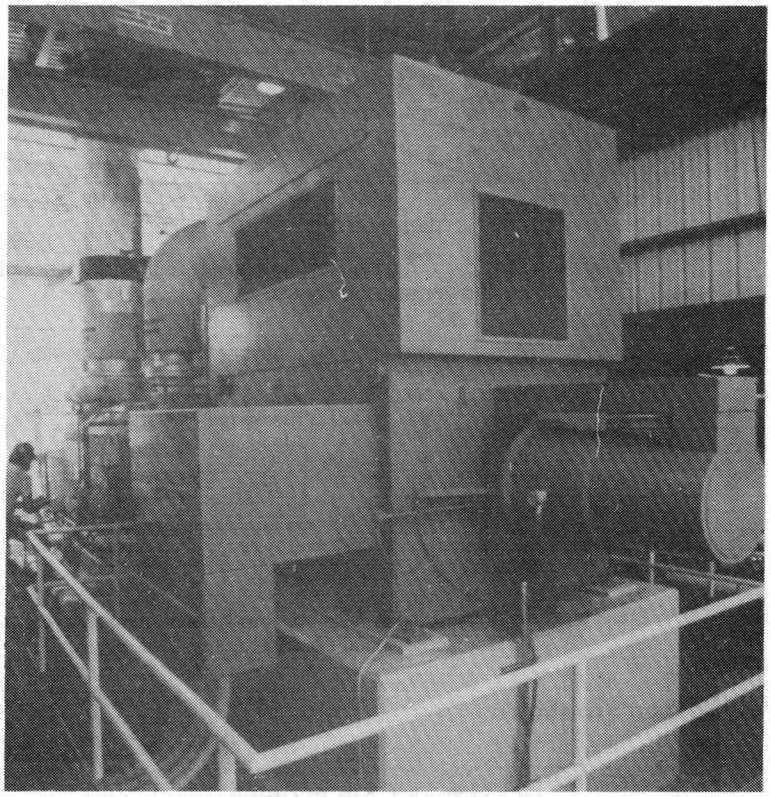

Fig. 23. Rear view of 10 000-hp drive motor.

To the foregoing gas horsepower must be added the friction horsepower consumed by bearings, seals, and gears. Fortunately, most compressor vendors with computer programs handy will provide a set of flow–pressure and flow–horsepower curves for a specific application similar to that shown in Fig. 21. This overcomes the long tedious calculations of working this out by hand computer. To these curves the user must add his system resistance curve.

From this point the format for evaluating adjustable-speed versus throttling control is much the same as it was for the pump example given earlier. As an example of this, a study was made on a 10 000-hp compressor in a petrochemical plant.

B. Example

The compressor and motor are shown in Figs. 22 and 23. The 8000 operating hours during the year were broken down into the following duty cycle:

2000 h @ 90 percent flow
6000 h @ 70 percent flow.

The system curve was determined and added to the compressor curves in Fig. 21 to be those shown in Fig. 24. Comparisons of various adjustable speed drivers to a base case of a fixed speed motor with throttling were made as shown in Figs. 25–29.

Space limitations do not permit the detailed calculations. The example chosen was taken from [8]. A summary of the study for a 15-yr life of the drive is as follows in Table IV.

The gross savings in Table IV would naturally be decreased by the time value of money. Fig. 30 gives a picture of the gross savings in the 15-year life of the plant by using

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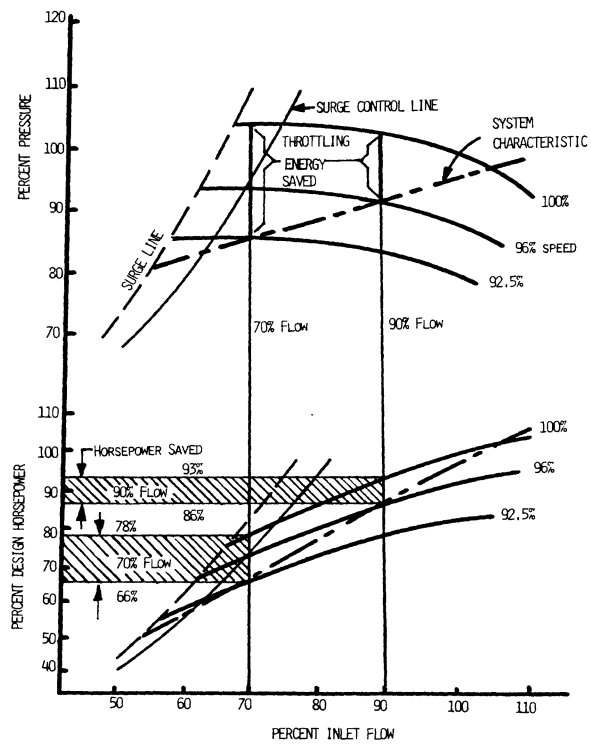

Fig. 24. Compressor and system curves at 70- and 90-percent flow.

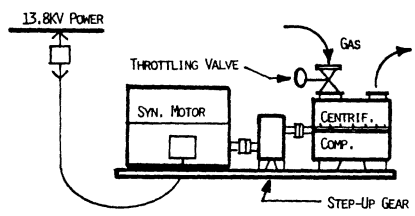

Fig. 25. Case I, fixed speed motor with inlet throttling.

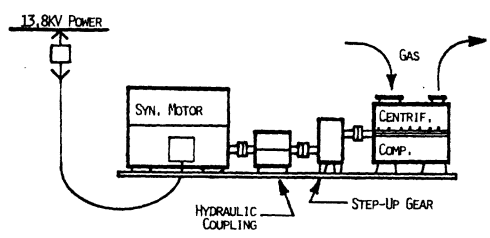

Fig. 26. Case IIA, fixed speed drive motor with hydraulic coupling.

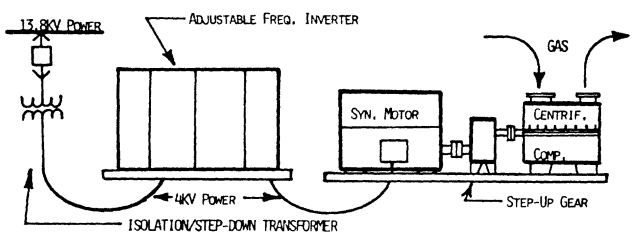

Fig. 27. Case IIB, adjustable frequency drive.

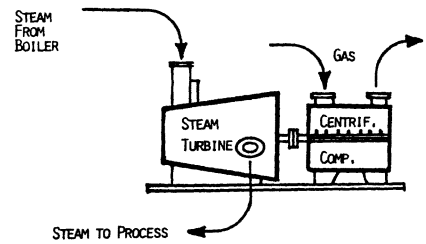

Fig. 28. Case IIC, noncondensing steam turbine drive.

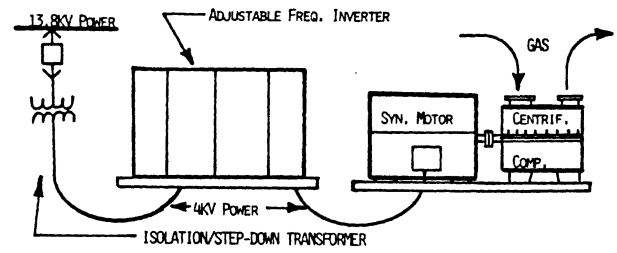

Fig. 29. Case IID, adjustable high-speed gearless drive.

TABLE IV
COMPARISON OF SAVINGS WITH VARIOUS DRIVES WITH COMPRESSOR

Case	Drive	Installed Cost	Energy Cost/Yr	Savings Over Fixed Speed/Yr	Gross Savings Over Fixed Speed
I	fixed speed/ throttling	$ 626 000	$2 603 000	$ ——	$ ——
IIA	hydraulic coupling	952 000	2 484 049	118 951	1 458 265
IIB	adjustable frequency	1 273 000	2 335 449	267 551	3 366 265
IIC	steam turbine	1 454 000	2 571 201	31 799	(351 015)
IID	gearless adjustable frequency	1 339 600	2 290 986	312 014	4 367 610

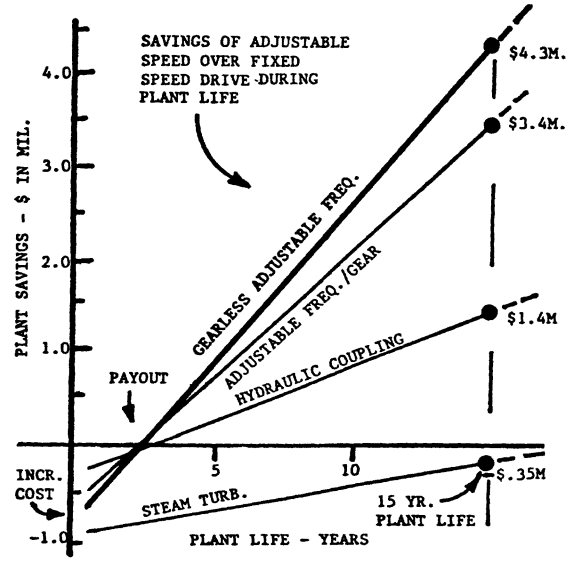

Fig. 30. Gross savings for 15-yr life of adjustable speed drives.

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adjustable speed in place of fixed speed with throttling. Though a gearless adjustable high-speed drive was not available at the time, it is added here for comparison.

OTHER FRINGE BENEFITS

Other benefits in using adjustable speed are not directly connected with savings energy. Some of these apply to motor drives only. Others are available with all types of drives. Though we have called them fringe benefits and they come free with the package of reasons for using adjustable speed, they can at times be the main reason for selecting adjustable speed. The dollar value of these additional benefits is hard to fix and is made mainly by user evaluation in a specific application. The next application may find them entirely different. A summary of main additional benefits follows.

A. Soft Start of Drive

Adjustable frequency drives inherently start with low inrush currents. If the driven load has an exponential speed–torque curve for starting, this is only 30–50 percent of full load current. If it requires constant torque, this may be as high as 100–150 percent full load current. Fixed speed drives and those driving hydraulic couplings may have 400–650 percent inrush current. With the exception of nonelectrical type drives such as steam and gas turbines, this has the following advantages.

1) Power System Reduced: Because of high inrush currents, power systems must be designed to hold their voltage up during starting. This is particularly important if one drive is much larger than the rest on a given utilization bus. To accomplish this with motor high starting inrush currents, the power system must be stiffened up at times with more than sufficient transformer capacity. This results also in switchgear interrupting duty higher than otherwise needed. Use of adjustable frequency drives can sometimes save money on the power system by using a system with lower starting capability. They are particularly suitable for starting on weak power systems, and at times, as in remote pump stations, this can be the major reason for their use. Note in passing that, as adjustable frequency drives do not provide a short circuit contribution to other faults on the system, this can be a further help towards reducing interrupting duty on the bus. One exception to this is when, in some types of inverters, the drive happens to be operating in a regenerating mode at the instant of fault.

2) Rotating Equipment Life Lengthened: Motor life is determined in part by the number of starts a motor makes. Normally, high locked rotor currents and the accompanying magnetic forces involved in fast and uncontrolled acceleration causes high temperatures, end turn coil movement, and uneven thermal expansion of rotors. Ultimately, the thermal and magnetic forces hasten failure. Larger motors have a limitation of only two starts per hour. In adjustable frequency drives with low starting currents, acceration time is controlled and is done very slowly. No limitations are placed on the number of starts per hour, and life is made much easier for the motor. This can extend motor life well over that of a fixed speed motor. Lack of high impact and transient torques also tends to increase the life of couplings and gear teeth in gears. Steam and gas turbines also provide the same benefit.

B. Increased Motor Life

Adjustable speed drives operate a good part of their life at something less than rated speed and power. As speed drops below rated value, bearing life tends to go up. This is particularly true in small drives with ball bearings where bearing life goes up by about the 1.6th power as speed drops. A drive operating at less than rated horsepower has its insulation temperature reduced and insulation life is approximately doubled for each 10° drop in temperature. The dc link in nearly all kinds of adjustable frequency and dc drives along with the isolation transformer isolate and shield the motor from power system problems. This may include low incoming voltage which is adjusted for in the control circuit. It can correct for unbalanced ac voltage on the power system which causes negative sequence torques and thus additional motor heating. Again, steam and gas turbines are naturally not bothered by power system disturbances.

C. Shortened Compressor Startup Time

Gas compressors pumping gas with less density than that of air, as say hydrogen, can frequently not be operated at rated speed unless that gas is available to be pumped. To use air would, say, require twice the horsepower. Fixed speed compressors which can only be operated at full speed cannot be started for checkout at installation until the rest of the process is complete and able to furnish the gas it will pump. With any type of adjustable speed drive, the compressor can be started and pretty well checked out and loaded with air until a horsepower limitation is reached. This advance checkout with air can save considerable project completion time, and in one case, up to a month in startup time.

D. Process Ride-Through

Fixed speed drives, operating when a power system disturbance such as a nearby fault occurs, cannot continue operation for more than a few cycles with no voltage. When voltage returns after the interruption, the drive has slowed to the point where it cannot reaccelerate the driven load unless the load is reduced immediately, as with automatic unloader valves on a compressor. Very few drives have these, and thus most drives will be tripped off by locked rotor protection when motors stall trying to reaccelerate at reduced torque while fully loaded. This is indicated in Fig. 31. By contrast, an adjustable frequency or dc drive has the capability to catch the drive when it is slowing down and under slow reacceleration with full load torque available bring the drive back up to its preset speed. This ride-through capability can save many process shutdowns from unexpected voltage disturbances on the power system. Adjustable speed drives with hydraulic couplings can accomplish the same thing if they can reduce speed quickly with the coupling. All drives must be caught and reaccelerated before so much drop in speed occurs that compressor seals are upset.

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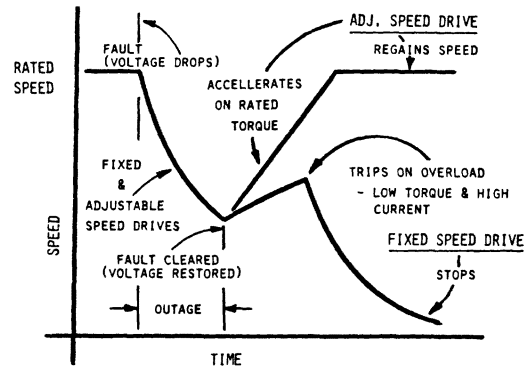

Fig. 31. Process ride-through on power outage for fixed speed and adjustable speed drives.

E. Increased Speed Range

Adjustable frequency drives do not know that 60 Hz is a standard frequency. Inverters can take their motors to well over their 60-Hz speed if needed. In some drives additional speed over rated is sometimes desired, and where horsepower rating is not exceeded, the capability to extend the speed range is available. For practical limits this may be only 10–15 percent more. Torsional problems on the motor and/or drive equipment may occur somewhere beyond that point. Since horsepower in many cases goes up as the cube of the speed, it is not long before a horsepower limit is reached. Ten to 15 percent extra throughput in most drives is considerable and often is a welcomed addition. This would not be available on a fixed speed or hydraulic coupling drive. A dc, steam, or a gas turbine could possibly provide this benefit also.

F. Extended Operating Range Before Surge Line

As illustrated in Fig. 32 a fixed speed pump or compressor as it is throttled back to reduce flow can only be reduced in flow so far before it encounters the surge line, a point below which operation is unstable and possible damage to bearings and seals is likely to occur. Because of control system speed drift or error or change in the line, usually a ten-percent margin of safety is held away from the line. As pressure is reduced for a given flow, the available control range is inherently increased as shown in the set of curves. The surge line goes down and to the left. Adjustable speed provides greater flow at less pressure and therefore a greater possible control range before the surge line is encountered. In Fig. 32 the fixed speed pump would have a usable operating range of 75–100 percent or only 25-percent range in flow. An adjustable speed pump would be able to utilize 30–100 percent or 70-percent flow range.

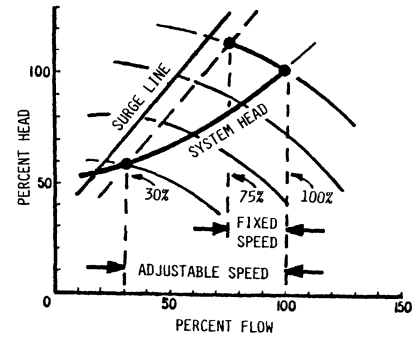

Fig. 32. Flow range in compressors and pumps is increased by adjustable speed over fixed speed.

CONCLUSION

An energy audit of process rotating equipment will show many cases where adjustable speed can save energy. Each case must be looked at carefully, knowing all application conditions such as:

1) driven equipment characteristic curve for the exact conditions on which it will operate;
2) flow duty cycle for the equipment in the process;
3) equipment and installation costs; and
4) per unit cost of energy, i.e., \$/1000 lb of steam, ¢/kWh, etc.

For best savings in losses, look for

- steep H–Q curve
- low static pressure,
- large duty cycle variations,
- large variation in flows,
- steep system head curve,
- system friction increases with age,
- a variety of fluid densities.

Savings on many energy saving projects will turn out to be great, while negligible on some. Those making decisions on whether to use adjustable speed over other forms of control should investigate the application completely and add up possible savings from drivers, from the driven rotating equipment, and from other fringe benefits. Adjustable speed is turning out to be a very profitable tool in today's search to find and reduce energy losses.

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Herbert N. Hickok (M'53–SM'83) was born in San Antonio, TX. He received the B.S.M.E. degree from the University of Texas, Austin.

He has been employed by the General Electric Company for 37 years, most of the time being spent as an Industrial Application Engineer. He works mainly with the petroleum, chemical, and electrocemical industries. His current position is that of Industry Engineer.

Mr. Hickok is a Registered Professional Engineer in the State of Texas. He has been active for many years in both local and national IEEE and ASME activities.

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